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CALCULATIONS FOR HRE NO. 2 HEAT EXCHANGER

By
C. L. Segaser

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Oak Ridge National Laboratory
Oak Ridge, Tennessee

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Work performed under Contract No. W-7405-Eng-26

OAK RIDGE NATIONAL LABORATORY
Operated By
CARBIDE AND CARBON CHEMICALS COMPANY
POST OFFICE BOX P
OAK RIDGE, TENNESSEE

Calculations have been made for the heat transfer surface area, number of tubes, length of tubes, hold-up volume in the tubes, and the pressure drop through the tubes for a proposed main heat exchanger for the HRE No. 2. In general, the calculations for the required surface area were made in accordance with the method described in CF-49-8-231 and CF-52-10-195.

Engineering design data on which the calculations were based were specified by W. R. Gall as follows:

Fuel Inlet Temperature	300^oC (572^oF)
Total heat	3000 kw
Tube size	3/8 in. OD (U-Tube)
Fuel	20 g/liter UO_2SO_4 in H_2O or D_2O
Fuel pressure	2000 psia
Flow rate	220 gpm
Steam pressure	520 psia

Supplementary design data required for the determination of the film coefficient in the tubes of the heat exchanger were obtained from CF-52-1-124 and are listed as follows:

Density of fuel solution	54 lb/ft^3
Specific heat of fuel solution	1.20 $Btu/lb\text{-}^oF$
Viscosity of fuel solution	0.270 lb/hr-ft
Thermal conductivity of fuel solution. .	0.350 $Btu/hr\text{-}lb\text{-}^oF/ft$

These properties of the fuel solution were evaluated at an average temperature in the heat exchanger tubes of 527^oF.

<u>The Heat Balance</u>

In order to determine the process temperatures at the entrance and exit of the heat exchanger tubes, a heat balance to transfer 3000 kw (1.025×10^7 Btu/hr) as sensible heat from the fuel solution to latent heat of vaporization

of water on the shell side at 520 psia was made.

$$Q = WC_p\,(T_i - T_o)$$

where, Q = Sensible heat given up by fuel, Btu/hr.

W = Flow rate of fuel, lb/hr.

C_p = Average specific heat of fuel, Btu/lb-$^\circ$F.

T_i = Initial or entrance temperature of fuel, $^\circ$F.

T_o = Final or outlet temperature of fuel, $^\circ$F.

Since the circulating pump is assumed to deliver a fixed quantity of 220 gpm of solution,

$$W = \frac{(220)(60)}{7.48}\,\rho = 1765\,\rho \text{ lb/hr.}$$

where, ρ = Average density of fuel, lb/ft^3.

therefore, $Q = 1765\,(\rho\,C_p)(T_i - T_o)$.

The product $(\rho\,C_p)$ must correspond to the average fuel solution temperature in the tubes, $\frac{T_o + T_i}{2}$. Figure 4 shows that the density of the solution decreases and the specific heat increases with temperature, but that the product $(\rho\,C_p)$ does not have such a wide variation, in fact, having a fairly constant value between the temperatures of $\sim 300^\circ$F and 500°F. By trial and error calculations, the temperatures at the entrance and exit of the tubes were determined such that the average temperature in the tubes coincided with an assumed value of the product $(\rho\,C_p)$. The results of this calculation are tablulated as follows:

$$T_i = 572^\circ F$$

$$T_o = 481.5^\circ F$$

$$T_{avg} = \frac{572 + 481.5}{2} = 527^\circ F$$

$$T_B = 471^{\circ}F \text{ (Saturation temperature of steam at 520 psia)}$$

$$T_i - T_o = 90.5^{\circ}F$$

$$T_i - T_B = 101^{\circ}F$$

$$T_o - T_B = 10.5^{\circ}F$$

<u>The Tubes</u>

The required wall thickness of a Type 347 stainless steel 3/8 in. OD tube at a design pressure of 2000 psia, a corrosion allowance of 0.030 in. and an allowable stress of 15,000 psi was calculated from the equation, as recommended in Section VIII of the 1952 edition of the ASME Unfired Pressure Vessel Code.

$$t = \frac{PR}{SE - 0.6P} + C$$

where t = thickness of wall, inches.

P = design pressure, psi.

R = inside radius of tube, inches.

S = allowable stress, psi.

E = joint efficiency ($E = 1$ for seamless tubes).

C = corrosion allowance, inches.

This equation shows that a 3/8 in. OD - 18 Ga (0.049 wt.) tube is good for only 1590 psi. Therefore, a 3/8 in. OD - 16 Ga (0.065 wt.) tube, good for $P = 3030$ psi is specified.

<u>Tube Data</u>

Flow area of 3/8" OD - 16 Ga(0.065 wt) tube	= 0.000327 ft^2
Outside heat transfer area per ft. of length	= 0.0982 ft^2
Thermal conductivity of Type 347 stainless steel	= 130 Btu/hr-ft^2-$^{\circ}F$/in.
Wall coefficient of 3/8" OD - 16 Ga tube	= 2000 Btu/hr-ft^2-$^{\circ}F$

Heat Transfer Surface Area

In CF-49-8-231 it was shown that the surface area of an exchanger, in which there is a substantial variation in U (the overall heat transfer coefficient) with temperature from one end of the tubes to the other, may be calculated from,

$$A = WC_p \int_{T_1 - T_B}^{T_o - T_B} \frac{d(T - T_B)}{U\ (T - T_B)}$$

Since, $T - T_B = \Delta t_i + \Delta t_w + \Delta t_o$

and $$\Delta t_i = \frac{\frac{Q}{A_i}}{h_i} = \frac{\frac{Q}{A_o}}{\left(\frac{D_i}{D_o}\right) h_i}$$

$$\Delta t_w = \frac{\frac{Q}{A_w}}{h_w} = \frac{\frac{Q}{A_o}}{\left(\frac{D_w}{D_o}\right) h_w}$$

$$\Delta t_o = \frac{\frac{Q}{A_o}}{h_o}$$

Substituting $$T - T_B = \frac{\frac{Q}{A_o}}{\left(\frac{D_i}{D_o}\right) h_i} + \frac{\frac{Q}{A_o}}{\left(\frac{D_w}{D_o}\right) h_w} + \frac{\frac{Q}{A_o}}{h_o}$$

where, A_o = outside tube surface area, ft^2

W = flow rate of fuel, lb/hr

C_p = specific heat of fuel, Btu/lb - °F

$T_o - T_B$ = overall temperature difference at cold end, °F

$T_i - T_B$ = overall temperature difference at hot end, °F

U = overall coefficient of heat transfer as a function of $T - T_B$, Btu/hr - ft^2 - °F

Δt_i = temperature drop thru inner film, °F

Δt_w = temperature drop thru tube wall, °F

Δt_o = temperature drop thru outer film, °F

D_i = inside tube diameter

D_o = outside tube diameter

D_w = average tube diameter

h_i = inner film coefficient, Btu/hr-ft^2-°F

h_w = wall coefficient, Btu/hr-ft^2-°F

h_o = outer tube wall coefficient, Btu/hr-ft^2-°F

The inside tube heat transfer coefficient, h_i, may be determined from the equation

$$\frac{h_i D_i}{k} = 0.023 \left(\frac{D_i V\rho}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{0.3}$$

which reduces to $h_i = 1.19 V^{0.8}$ when the average properties of the fuel solution are substituted. In this equation, V is the bulk velocity through the tubes in ft/hr.

The boiling side coefficient may be calculated from data taken from Figure 156 of Heat Transmission by McAdams corrected for the specified steam pressure of 520 psia. Between the limits of heat transfer by natural

convection represented by the region AB' of Figure 147 (McAdams) and the region of peak flux at a Δt_o of 45°, the curve on logarithmic paper is practically straight and the boiling side heat transfer coefficient may be expressed with reasonable accuracy by the equation

$$h_o \text{ (520 psi)} = 100\, \Delta t_o^{1.42}$$

Substituting these values for the individual film coefficients, the overall temperature gradient at any point along the tube becomes

$$T - T_B = \frac{\frac{Q}{A_o}}{\left(\frac{D_i}{D_o}\right)\left(1.19V^{0.8}\right)} + \frac{\frac{Q}{A_o}}{\left(\frac{D_w}{D_o}\right)2000} + \frac{\frac{Q}{A_o}}{100\,\Delta t_o^{1.42}}$$

or, since $\frac{Q}{A_o} = h_o\,\Delta t_o$

$$\Delta t_o = 0.149\left(\frac{Q}{A_o}\right)^{0.414}$$

hence, $$T - T_B = \frac{1}{0.778V^{0.8}}\;\frac{Q}{A_o} + \frac{1}{1650}\;\frac{Q}{A_o} + 0.149\left(\frac{Q}{A_o}\right)^{0.414}$$

Differentiating this equation, substituting in the integral equation for surface area and integrating between the limits of heat flux at the hot end to heat flux at the cold end; the following equation was derived for calculating the heat transfer area

$$A = WC_p\left[\frac{1}{0.778V^{0.8}}\;\ln\frac{\left(\frac{Q}{A_o}\right)_i}{\left(\frac{Q}{A_o}\right)_o} + \frac{1}{1650}\;\ln\frac{\left(\frac{Q}{A_o}\right)_i}{\left(\frac{Q}{A_o}\right)_o} - 0.105\left[\left(\frac{Q}{A_o}\right)_i^{-0.586} - \left(\frac{Q}{A_o}\right)_o^{-0.586}\right]\right]$$

where, A = Outer tube surface area, ft^2

W = Weight rate of flow thru tubes, lb/hr

C_p = Average specific heat of fuel, Btu/lb $^\circ F$

$\left(\frac{Q}{A_o}\right)_i$ = Heat flux at hot end, Btu/hr-ft^2

$\left(\frac{Q}{A_o}\right)_o$ = Heat flux at cold end, Btu/hr-ft^2

V = Velocity thru tubes, ft/hr

Thus, for a given flow rate and steam pressure, this equation shows that the surface area varies inversely as the 0.8 power of velocity.

Number of Tubes

For a given flow rate of 220 gpm, the number of tubes required will vary inversely as the velocity through the tubes. A flow rate of 220 gpm is equivalent to 0.491 ft^3/sec. Since the flow area per tube is 3.27 x 10^{-4} ft^2,

$$0.491 = 3.27 \times 10^{-4} NV$$

and

$$N = \frac{0.491}{3.27 \times 10^{-4} V}$$

Average Length of Tubes

Multiplying the outside surface area per foot of length, by the average length per tube and the total number of tubes will give the surface area as shown by the following equation

$$S_T = 0.0982 NL$$

From which,

$$L = \frac{S_T}{0.0982 N}$$

Hold-up Volume of Tubes

In a similar manner, the hold-up volume in the tubes may be shown to be

$$V = 3.27 \times 10^{-4} NL \text{ (Ft}^3\text{)}$$

Pressure Drop Through Tubes

The pressure drop through the tubes was calculated by the familiar Fanning equation,

$$\Delta P_F = \frac{4f\rho \, V^2 L}{2g \, D_i}$$

Table 1

Tabulated Data for HRE No. 2 Heat Exchanger

		Velocity thru tubes, ft/sec				
		5	10	15	20	25
Flow Rate, gpm		220	220	220	220	220
Power (3000 Kw)		← 1.025×10^7 Btu/hr →				
Temperature	T_i, °F	572	572	572	572	572
	T_o, °F	481.5	481.5	481.5	481.5	481.5
Surface Area (ft^2)	Clean	404	338	314	298	290
	Fouled	540	450	418	398	387
LMTD °F		40.1	40.1	40.1	40.1	40.1
Apparent U Btu/hr-ft^2-°F	Clean	633	760	814	862	883
	Fouled	473	570	610	646	662
Number of Tubes		300	150	100	75	60
Length of Tubes, ft	Clean	13.75	23.0	32.0	40.5	49.5
	Fouled	18.3	30.6	42.7	54.0	66.0
Hold-up in Tubes (Liters)	Clean	38.2	31.8	29.8	28.0	27.4
	Fouled	51	42.5	39.7	37.4	36.6
Pressure drop, psi	Clean	1.95	11.20	32.8	68.2	123.0
	Fouled	2.60	15.0	43.7	91.0	164.0

Results of Calculations

The heat transfer surface area based on the outside diameter of the tubes, the number of tubes, the length of the tubes, the hold-up volume and the pressure drop through the tubes have been calculated for fluid velocities from 5 ft/sec through 25 ft/sec using 3/8" OD - 16 Ga (0.065 wt) Type 347 stainless steel seamless tubes for a 3000 Kw vaporizing exchanger generating saturated steam at 520 psia pressure. The results are listed in Table 1. These results have been tabulated for both clean surface and fouled surface, where the fouled surface is assumed to be 4/3 of the clean surface. The results for the fouled surface account for scale build-up on both the inner and outer surfaces of the tubes and provide a factor of safety for specifying the exchanger.

Where an exchanger has a constant U, it is customary to calculate the surface area from the equation,

$$Q = UA\Delta T$$

where ΔT is defined as the logarithmic mean temperature difference as calculated from

$$\Delta T = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

Occasionally where there is a substantial variation in U, this method is still used. As a matter of record, some of the heat exchanger manufacturers contacted by ORNL have apparently used this method for specifying vaporizing heat exchangers for the ISHR using a constant U = 750 Btu/hr-ft^2-$^\circ$F. This coefficient was used for calculating the fouled surface area, based on 3/4 of an assumed U = 1000 Btu/hr-ft^2-$^\circ$F for clean surfaces. For purpose of comparison, the apparent overall U for the HRE No. 2 exchanger is shown.

This U was calculated for both the clean surface and the fouled surface using a logarithmic mean temperature difference based on the entrance and exit process temperatures and the saturated steam temperature. The LMTD for this exchanger is 40.1°F, hence, the apparent U is,

$$U = \frac{Q}{40.1\ A_o}$$

The results from Table 1 have been plotted on Figures 1 through 7 for ease in interpolating data for any velocity from 0 up to 25 ft/sec.

Figure 1 shows the required surface areas for both the clean and fouled tube surfaces.

Figure 2 is a plot of heat flux, Q/A_o, versus overall temperature drop $T - T_B$ for velocities from 5 through 25 ft/sec.

Figure 3 shows the variation in outside heat transfer coefficient as a function of temperature difference between the tube surface and the saturated liquid for pressures of 14.7 psia, 215 psia, 370 psia and 520 psia, from horizontal tubes.

Figure 4 shows the variation of density, specific heat and the product of the two at increasing temperatures for a 20 g/l solution of UO_2SO_4 in D_2O.

Figure 5 gives the number of tubes and average length of tubes as a function of velocity for the exchanger.

Figure 6 shows the decrease in hold-up required as the velocity through the tubes is increased.

Figure 7 shows that the pressure drop through the tubes of the heat exchanger increases very rapidly with velocity and becomes exorbitantly high at 25 ft/sec.

Preliminary Recommendations

A meeting was held in the Conference Room of Building 9204-1 on Friday, December 11, 1953, between members of the REED Design, Development, and Operation Sections to discuss a proposed design for the HRE No. 2 Main Heat Exchanger. Those present were

Design	Development	Operation
W. R. Gall	C. B. Graham	S. E. Beall
R. B. Briggs	W. L. Ross	
W. Terry	L. F. Goode	
C. L. Segaser		

(1) The calculations presented in the enclosed memorandum were discussed, and a sketch of a proposed heat exchanger made by W. Terry based on a velocity of 15 ft/sec through 100 3/8" O.D. - 16 Ga. Type 347 stainless steel tubes was shown. At 15 ft/sec the average length of tubes required is 32 ft. The exchanger surface shown was contained in a shell approximately 30-inches diameter by 15-ft long. This results in an exchanger shell which has a length to diameter ratio of 6:1, and the decision was somewhat arbitrarily made to reduce the velocity from 15 ft/sec to 10 ft/sec. This will result in increasing the number of tubes from 100 to 150 and decreasing the tube length from 32 ft to 23 ft with a consequent lower pressure drop and a better proportioned heat exchanger shell, but at the expense of approximately 2 liters of greater hold-up volume.

(2) R. B. Briggs recommends that the surface area of the exchanger should be specified on the basis of the clean surfaces rather than the fouled surfaces. He bases this opinion on the performance of the existing HRE No. 1 heat exchanger which was specified neglecting scale build-up and which has apparently functioned satisfactorily during the relatively short time it has been operated.

(3) The specifications of a vaporizing heat exchanger for the HRE No. 2 based on the above premises then may be tentatively listed as follows

Fuel	20 g/1 UO_2SO_4 - D_2O or H_2O
Fuel Pressure (inlet)	2000 psia
Heat transfer	3000 kw
Tube size	3/8" OD - 16 Ga (0.065)
Flow rate	220 gpm
Steam pressure	520 psia
Steam temperature	471 ^{o}F
Fuel inlet temperature	572 ^{o}F
Fuel outlet temperature	481.5 ^{o}F
Fuel velocity	10 ft/sec
Heat transfer surface area	338 ft^2
Number of tubes	150
Average length of tubes	23 ft
Hold-up volume in tubes	31.8 liters
Pressure drop thru tubes	11.2 psi

(4) These specifications are based on the assumption that the bulk temperature of the boiler water is already at the saturation temperature of 471^{o}F corresponding to the generated steam pressure of 520 psia. Actually, feedwater returned from the steam system will be considerably less than this temperature and means must be provided to preheat this water to saturation temperature. In normal power plant practice, part of this preheat is provided by external feedwater heaters using steam bled from various stages of the turbine as the heating medium. In the HRE No. 1, boiler feedwater heat is provided in part from the D_2O cooler in the blanket system, and the

remainder is supplied through a feedwater heating element built into the vapor space of the main heat exchanger. Also, in the HRE No. 1 heat exchanger twelve additional tubes were provided over the calculated number as a safety factor to account for contingencies such as the above.

(5) The fabrication problems of the heat exchanger were discussed and the following decisions were tentatively proposed.

(a) The tube-sheet to be of solid construction of Type 347 stainless steel with the thickness determined from the TEMA Standard Procedures using an allowable design stress of 15,000 psi.

(b) The problem of leak-detection versus no leak-detection was presented. This problem is to be investigated further before a final decision is made.

(c) The tube bundle does not need to be removable from the shell.

(d) An efficient entrainment separator is desired for high quality steam.

(e) A closure design is desired which may be removed prior to final installation, but which may be seal welded before the reactor system is made critical.

(f) Feedwater surface area will be required for boiler water preheating.

(g) The feedwater temperature leaving the steam killer should be investigated.

(h) A delivery date of January 1, 1954 is desirable.

(i) Type 304 L stainless steel was discarded as a possible material of construction in favor of Type 347.

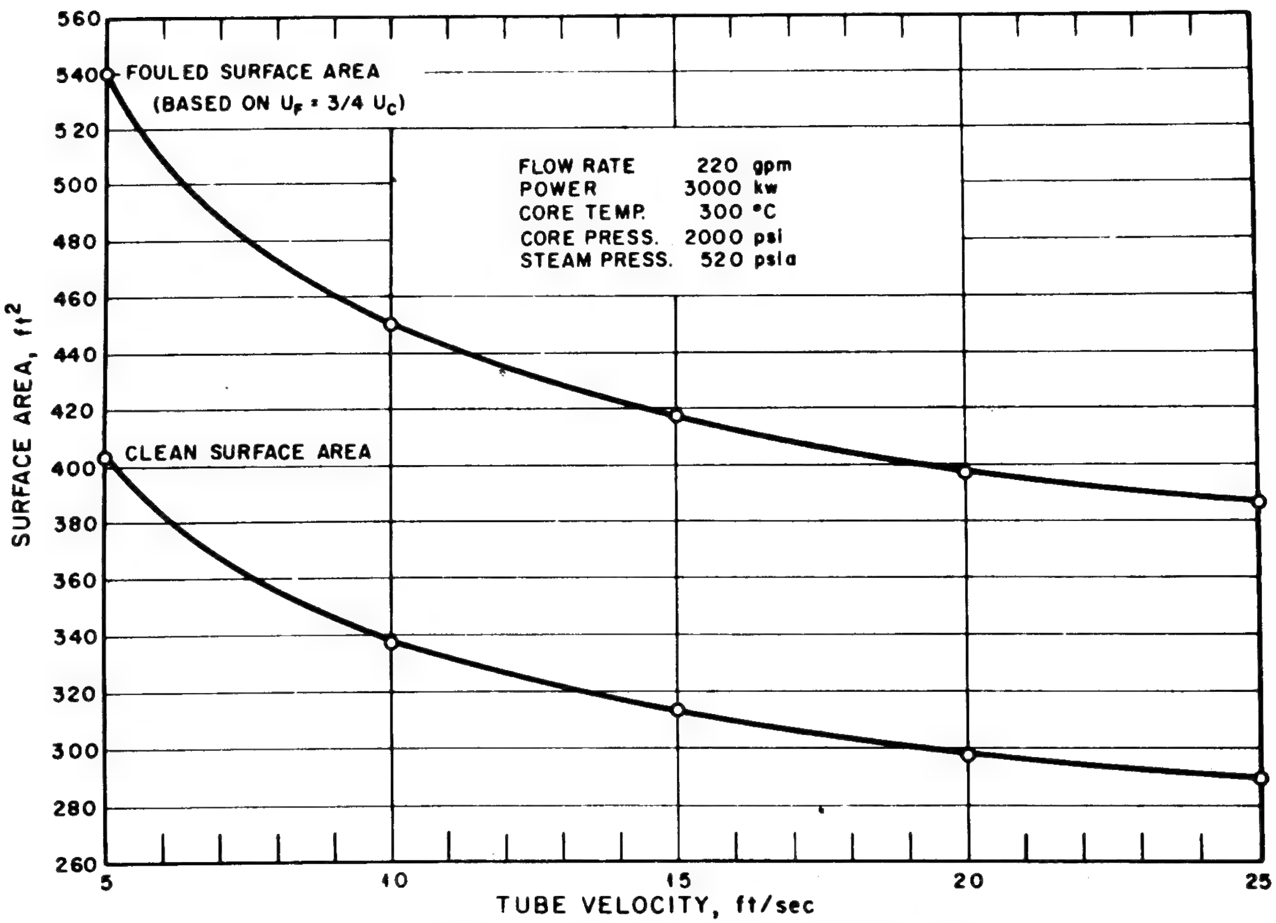

FIG. 1. HRE No. 2 HEAT EXCHANGER SURFACE AREA

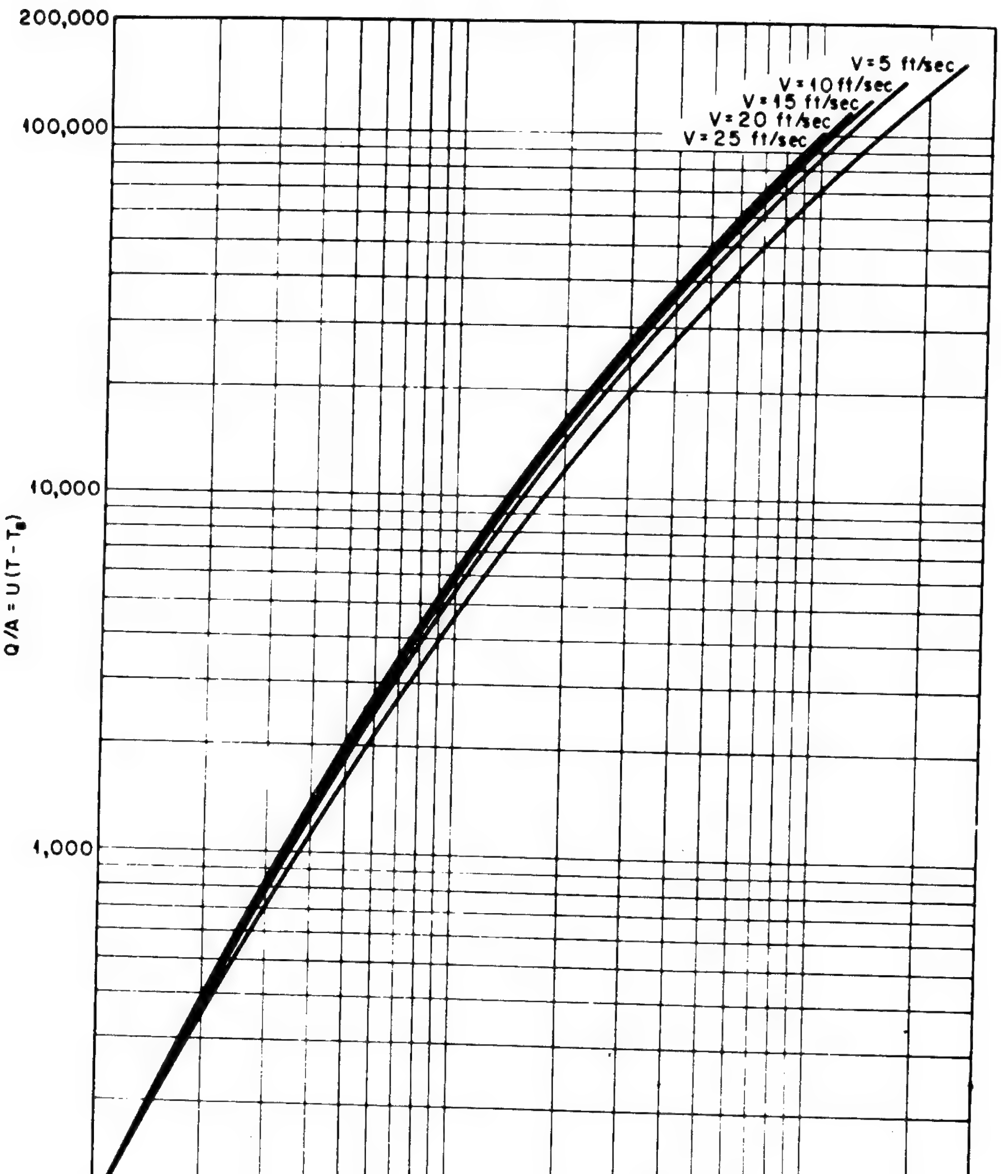

FIG. 2. FLOW RATE OF 220 GPM; STEAM PRESSURE = 520 psia

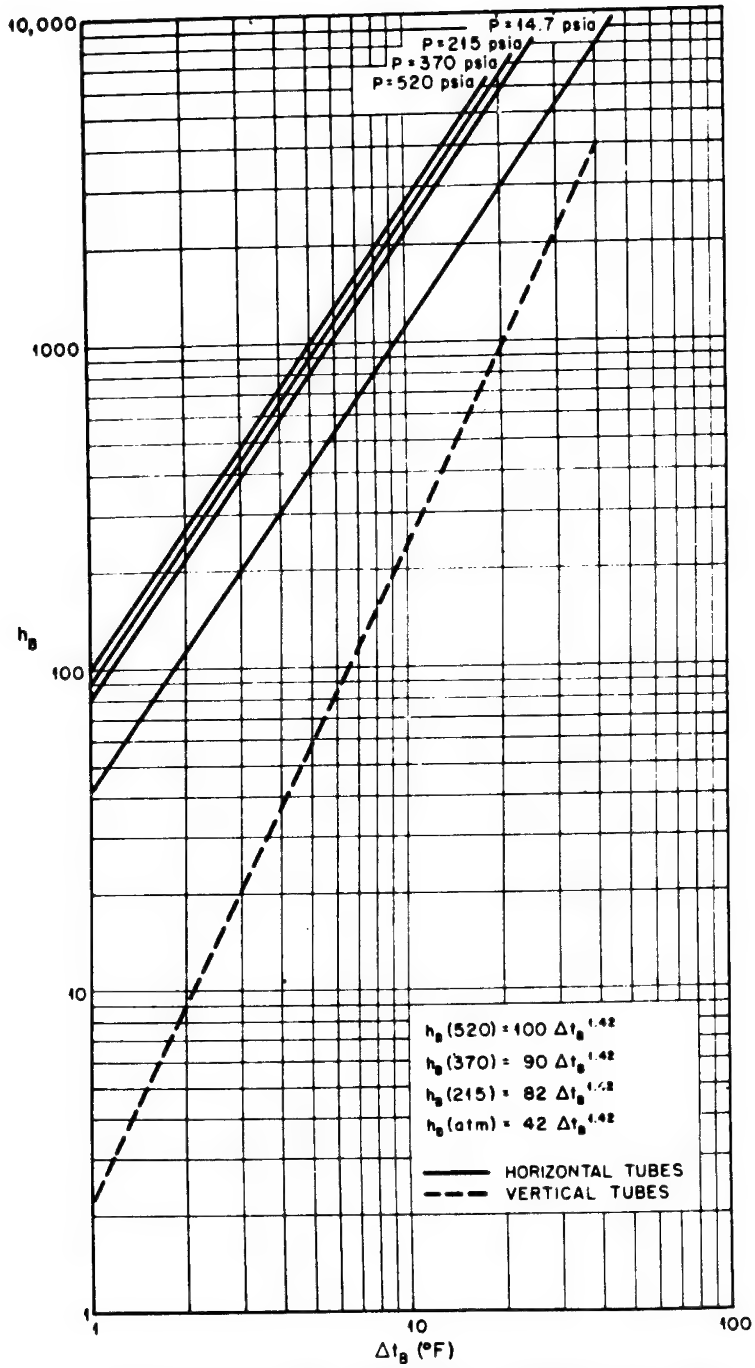

FIG. 3. BOILING SIDE COEFFICIENT – HORIZONTAL TUBES

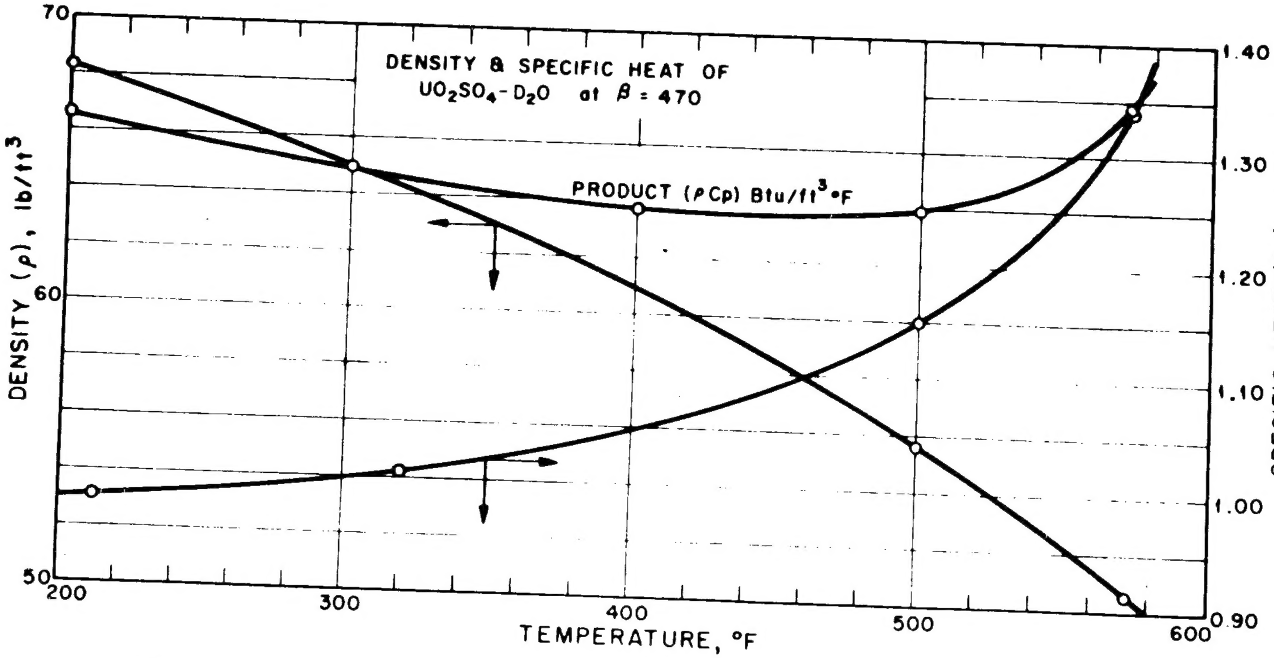

FIG. 4. FUEL SOLUTION PROPERTIES FOR $UO_2SO_4-D_2O$ AT β = 470

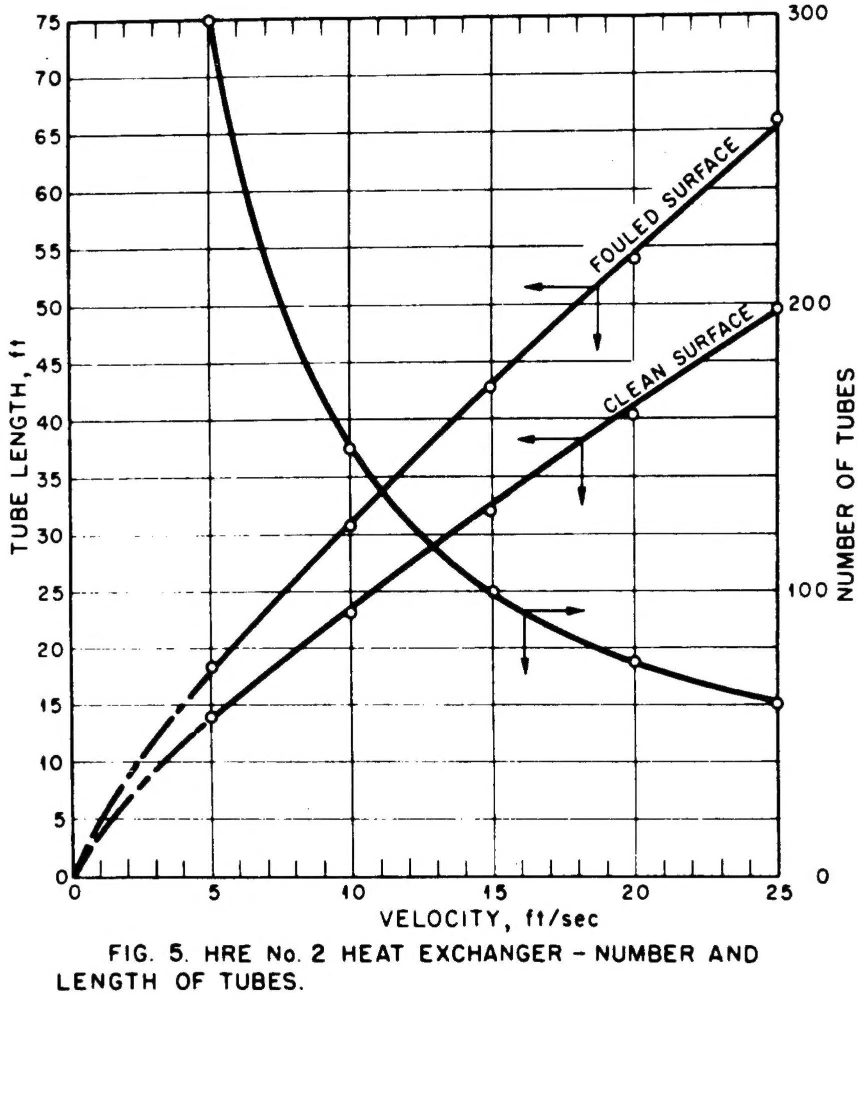

FIG. 5. HRE No. 2 HEAT EXCHANGER – NUMBER AND LENGTH OF TUBES.

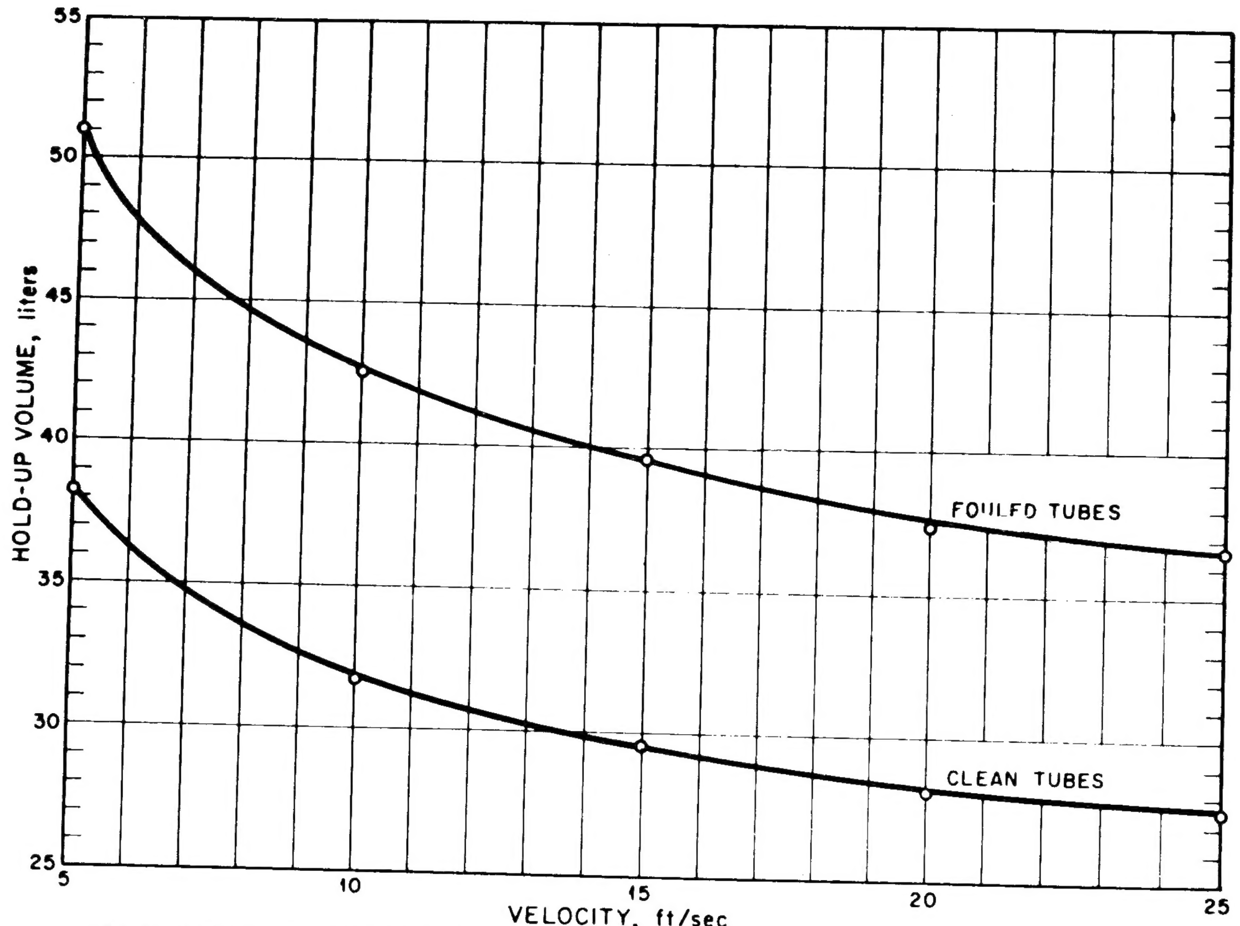

FIG. 6. HRE No. 2 HEAT EXCHANGER - HOLD-UP VOLUME IN TUBES

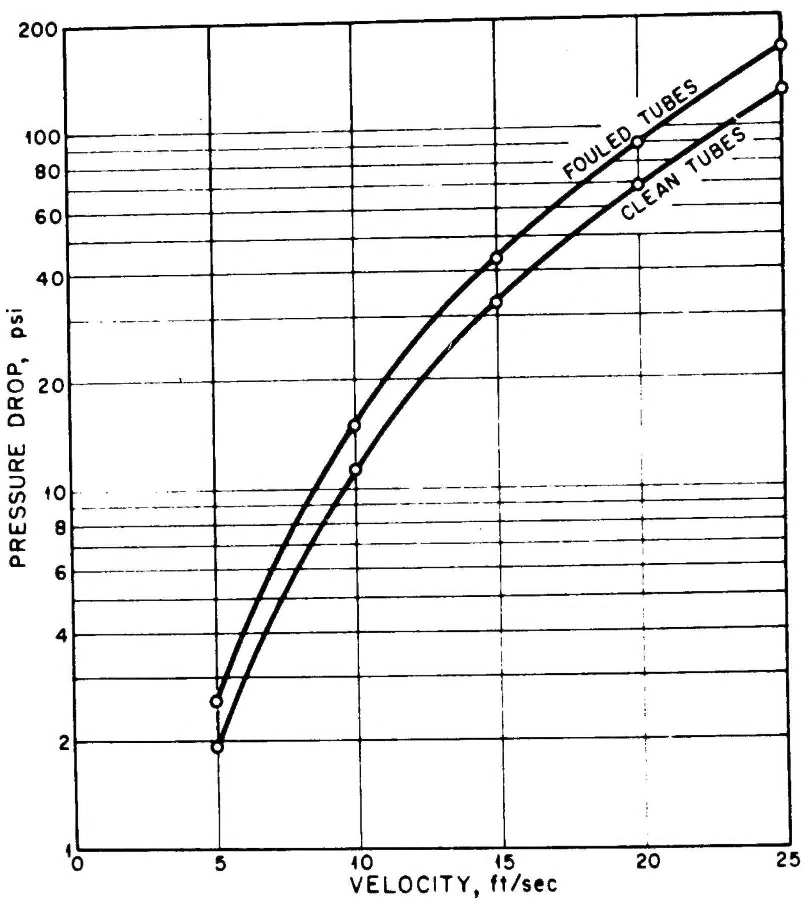

FIG. 7. HRE No. 2 HEAT EXCHANGER – PRESSURE DROP THROUGH TUBES.